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NASA Contractor Report 165473

(NASA-CR-165473) EVALUATION OF THE POTENTIAL OF THE STIRLING ENGINE FOR HEAVY DUTY APPLICATION Final Report (Stirling Thermal Motors, Inc.) 62 p HC A04/MF A01 CSCL 10B G3/44

N82-10505

Unclas 27762

EVALUATION OF THE POTENTIAL OF THE STIRLING ENGINE FOR HEAVY DUTY APPLICATION

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October 1981

Prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Under Contract NAS3-22226

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Section 1

INTRODUCTION - GOALS AND OBJECTIVES

Most of the publicly sponsored Stirling development work has concentrated on special light duty applications such as automotive propulsion. Information available to the public is, therefore, rather limited to such light duty applications of the Stirling engine. These light duty applications impose severe constraints on the freedom of the designer to realize all the benefits of the inherent flexibility and high efficiency of the Stirling cycle.

Engines designed to fulfill severe limitations on size, weight, starting time and difficult duty cycles may not be representative of the Stirling engine in general. They may be more complicated and expensive for large volume production than what they would be without the limitations of light duty applications.

Therefore, it is of value to determine the potential efficiency, cost and suitability to large volume production of a heavy duty Stirling engine (HDSE). This study addresses itself to this question and considers two cases:

1) Existing and near term technology, and

2) Advanced technology applied.

In order to assess the potential of the Stirling engine for heavy duty application the following approach was adopted:

- Requirements and features of the heavy duty application were identified.
- Conceptual design of an HDSE meeting the requirement identified in the preceding step was generated. This was done with the aim of achieving high efficiency, low cost and simple design, while applying only existing and near term technology.
- The conceptual HDSE was simulated yielding its performance characteristics.
- Areas where advanced technologies may bring about further improvements were identified and their potential was quantitatively estimated.

The results of this effort are described in the following sections.

Section 2

APPLICATION FEATURES AND REQUIREMENTS

The term heavy duty commonly applies to machinery designed for long life, low maintenance and for exposure to severe environmental conditions such as dirt and dust, humidity, extreme temperature and mechanical shocks.

In 1978, a total of approximately 24 million internal combustion engines were produced in the U.S.A. (excluding aircraft and outboard engines). Of these, 9.4 million were automotive and over 14 million were non-automotive. By far the largest population is the gasoline powered engine ranging in size from the fractional horsepower single cylinder to the multi-cylinder engines exceeding 1,000 hp. Diesels rank second with a range in sizes from about 5.0 hp to 55,000 hp.

The 14.4 million non-automotive internal combustion engines produced in 1978 fall into a wide range of sizes and types (figure 1) to fit the many requirements of the multiplicity of applications of the varied market segments (figure 2). There are about 12.4 million engines in the under 11 horsepower category and nearly another one million in the 11 to 30 horsepower range. They account for approximately 94% of the non-automotive internal combustion engine production. The diesel engine accounts for, roughly, another 2.5%, the remainder of the non-automotive markets. This

Figure 1 - Non-automotive internal combustion engine production - 1978

TOTAL PRODUCED (USA) - 14,131,608

NUMBER OF ENGINES	NUMBER OF CYLINDERS	TYPE	% OF TOTAL
13,245,778	1	Gasoline	93.7
220,128	2	Gasoline	
110,690	4	Gasoline	3.4%
38,384	6	Gasoline	
114,211	8 to 16 (95% were 8)	Gasoline	
31,596	2	Diesel	
106,068	4	Diesel	2.4%
205,100	10 to 20 (70% were 12)	Diesel	
14,071,955			99.5% *

* The balance is comprised of gas engines and a small quantity of miscellaneous engine sizes/types.

Source: Wards Automotive Yearbook, 1978

Figure 2 – Non-automotive I.C. engine market segments and applications general list

MARKET SEGMENT	MARKET APPLICATIONS
Air compressors	Compressors
Agricultural	Irrigation sets
	Agricultural tractors
	Combines
	Swathers
	Balers
	Mowers
	Sprayers
	Other agricultural equipment
Construction	Mining equipment
	Forest harvesting equipment
	Skidders
	Cranes
	Excavators
	Scrapers
	Graders
	Crawler tractors
	Rubber tired dozers
	Rubber tired loaders
	Paving equipment
	Boring & drill rigs
	Construction equipment
	Skid steer loaders
	Rollers & compactors
	Off highway trucks
	Asphalt pavers
	Trenchers
	Backhoes
	Cement & mortar mixers
	Concrete buggies
	Tampers
	Concrete finishers
	Concrete vibrators
General industrial	Pumps
	Oil field equipment
	Highway & railroad refrigeration
	Locomotives
	Scrubbers & sweepers
	Chippers
General industrial (continued)	Agg. Crushing & Proc. equipment
	Industrial tractors
	Other gen. industrial equipment
Highway	Pick-ups & vans (Class 1 & 2)
	Lt. commercial vehicles (Class 1 & 2)
	Trucks (Class 5)
	Trucks (Class 6)
	Trucks (Class 7)
	Trucks (Class 8)
	Buses
Lawn & Garden	Blowers & leaf vac.
	Trimmers & brush cutters
	Snowblowers
	Tillers
	Lawn & garden tractors
	Lawn mowers
	Chain saws
	Snowmobiles
	Wood splitters
	Other lawn & garden equip.
Marine-commercial	Marine-commercial
Material Handling	Fork lifts
	Other material handling equip.
	Tow tractor
	Terminal tractors
	Aerial lift
Marine pleasure	Marine pleasure
Other	Distributor repower
	Loose engine exports
	Tactical military vehicles
Welders/generators	Generator sets
	Welders
	Lighting plants

represents the heavy duty applications portion of the marketplace. It is interesting to note that even though the diesel engine (heavy duty) market represents less than 3% of the total number of non-automotive engines produced (see figure 1), it accounts for nearly 50% of the dollar volume.*

Since the purpose of this study is to investigate the potential of the Stirling engine for heavy duty applications, automotive engines will be excluded. The small light duty engine market shall also be excluded from consideration since the primary requisites for successful penetration into this marketplace are low cost and light weight. Low fuel consumption and long term operation at high load factors are not important in this marketplace.

An examination of the non-automotive diesel market reveals that over 90% of the production falls below 350 horsepower and over 70% below 200 horsepower. The largest segment of the market lies in the 100 to 200 horsepower category which accounts for approximately 40% of the production (figure 3). Therefore, it can be concluded that a viable size for the heavy duty Stirling engine lies in the 100 to 200 horsepower range. The demand for more fuel efficient engines has shown a steady increase over the last 5 years. However, the recent rapid increase in fuel costs has not resulted in either commercial or

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*Reference: U.S. Department of Commerce Industrial Outlook 1979

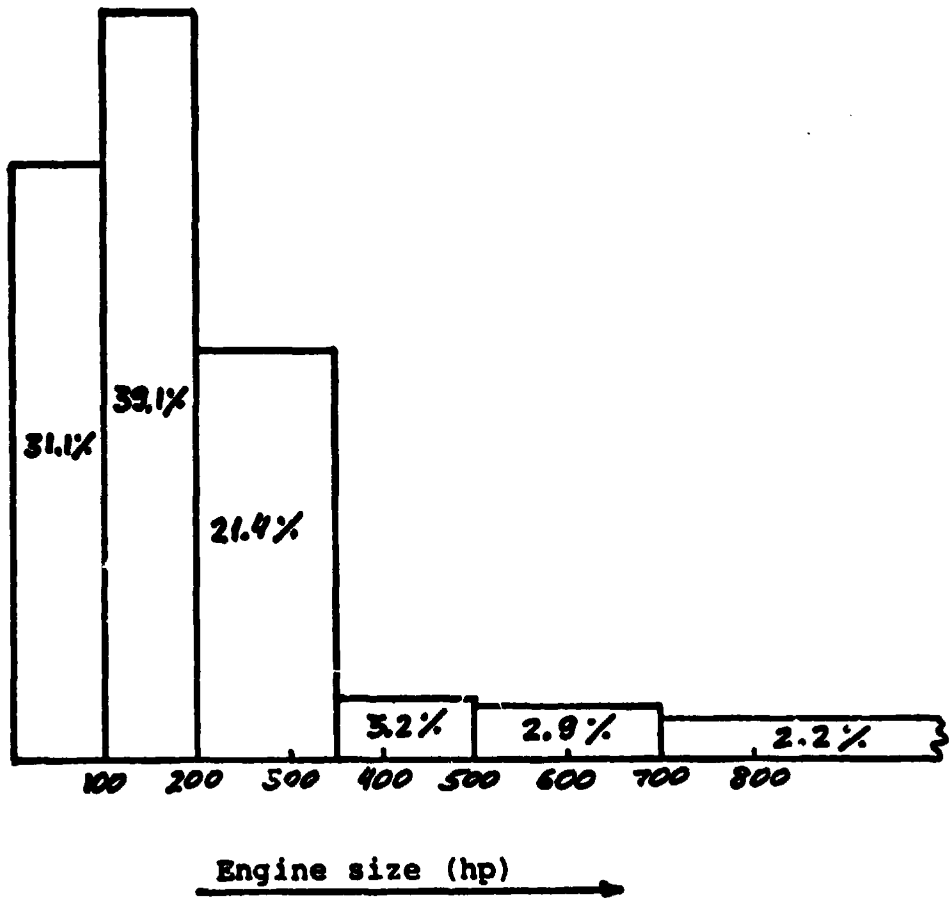


Figure 3 - 1978 Non-automotive diesel engine production relative size distribution

Source: Standard Industrial Classification (sic) Data (Dept. of Commerce) Diesel Progress 1979

industrial demands for heavy duty engines of smaller capacity, as is the case with automobiles, because sizes are selected for optimum match to power demand and duty cycle requirements. Therefore, the parameters for the heavy duty Stirling engine will be derived from the basic characteristics of diesel engines in this horsepower range.

Heavy duty engines are used in virtually every market segment. There is a complete range of sizes and types to fit practically every power requirement. The differences are found mainly in application related parameters influenced by the engine's speed; i.e., size and weight. From these parameters, engines can be classified into three basic categories:

1) Low speed engines with rpms of less than 350, which weigh between 40 and 50 pounds per horsepower.
2) Medium speed engines of 350 to 2,000 rpms, which weigh between 10 and 35 pounds per horsepower.
3) High speed engines with a speed range of over 2,000 rpms which weigh between 4 to 10 pounds per horsepower.

Development for military and special purposes, such as tanks and boats, can achieve power to weight ratios of less than 4 pounds per horsepower.

Heavy duty engines are manufactured to meet the specifications of the buyer. The marketing of industrial engines is unique in that the largest portion of the output

is sold to producers of other equipment and is purchased by the end user as a component of some other product or system such as a generator, pump, compressor, tractor, etc.

Based on the foregoing, a Stirling engine of 150 hp with features and requirements characterizing non-automotive diesel engines of that size has been selected for this study.

Such requirements and features include:

- Long service life (10,000 hours at full load)
- Resistance to adverse environmental conditions
- Low maintenance expectation
- Operational safety
- Variable load operation with rapid power variation
- Weight under 10 pounds per horsepower

To assure that the full potential of the HDSE is reflected in the conceptual design, the chosen design philosophy was to emphasize the following:

- High efficiency
- Low material cost
- Material availability
- Low production cost (in quantity)

Even though diesel engines can be started rapidly, the heavy duty application does not pose this requirement. It was, therefore, relaxed in order to allow the use of a heat pipe with the HDSE. In addition to improving the

efficiency, cost and material availability, it endows the HDSE with an additional feature:

- Multi-fuel capability

Service life is determined by creep of the hot stressed metal parts. No major maintenance operations are required during the 10,000 hours service life.

To insure compatibility with standard driven equipment of 150 hp a speed in the order of 1000 rpm is called for. With this speed a weight of 10 lbs/hp is achievable with a Stirling engine. Obviously, higher efficiency may be achieved at much lower speed with the engine still not heavier than a comparable diesel ($\sim$25 lbs/hp). This, however, will necessitate the use of very large, heavy and inefficient driven equipment or a 150 hp gear box.

Section 3

ENGINE CONCEPT USING EXISTING TECHNOLOGY

The engine that was conceptually designed in order to assess the potential of existing technology applied to a HDSE is described in this section. The philosophy and approach are outlined, the design implementing them is described and the resulting expected performance is presented. In order to realistically assess the influence of the engine size on the potential of HDSE's, the design concept was applied also to a large engine rated at 500 hp. The results of this effort are presented and compared with the 150 hp engine concept.

3.1 PHILOSOPHY AND APPROACH

The considerations yielding major design features are presented in this section. Included are these involved in the choice of the following features:

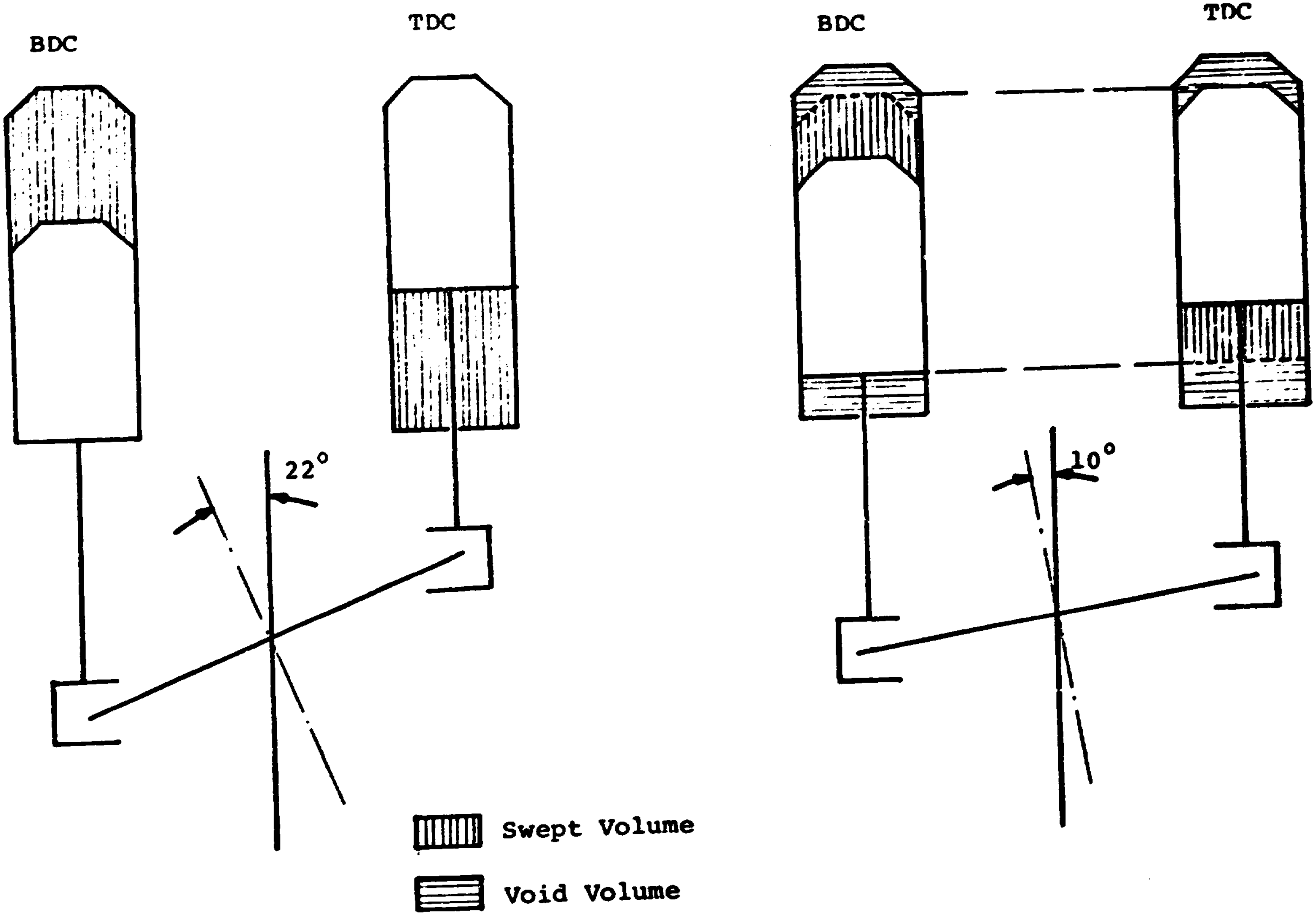
- configuration
- drive and power control
- working fluid
- materials
- relative magnitude of losses

3.1.1 Configuration, Drive and Power Control: The configuration chosen is a four cylinder, double acting engine with

a variable angle swashplate drive. The swashplate angle determines the piston stroke and thus the swept volume and hence the power. The drive and power control (which form a single system) is simple, compact and reliable, in particular when compared to the cumbersome and complicated mean pressure control. The control method, in addition to accommodating rapid torque variations, enhances the engine's part load efficiency: when the stroke is reduced a volume that has previously been swept by the piston becomes an additional void volume (see figure 4) which is instrumental in reducing the amplitude of the pressure wave and with it a number of significant losses.

The drive is also amenable to dynamic balancing, alleviating the need of a heavy mounting frame and enhancing component reliability. The compact drive and power control fits into a relatively small, roughly spherical crankcase that can, with only a small weight penalty, be kept at mean cycle pressure. This makes it possible to avoid the complicated and difficult problem of reciprocating rod sealing. A single commercially available rotating shaft seal will solve the working fluid containment problem and metal oil scrapers on the piston rods will prevent oil from migrating into the working spaces.

The double acting multi-cylinder configuration, chosen partly because it accommodates the variable swashplate



<u>Figure 4</u> - Combination of swept volume and void volume control by stroke variation with a variable swashplate

drive and power control with all its attractive features, offers a number of potential benefits: with four double acting cylinders the torque variations during a cycle are extremely small (similar to a 16 cylinder spark ignited I.C. engine). This eliminates the need for heavy flywheels. Finally, the multi-cylinder engine will have a higher specific power than the second best candidate - a single cylinder engine.

<u>3.1.2 Combustion and Heat Transport System</u>: It is strongly suggested that whenever possible, a remote heat source with a sodium (or other suitable liquid metal) heat pipe be used in conjunction with a Stirling engine. This offers a number of major advantages to the combination and makes the Stirling engine very attractive for applications where a heat pipe may be used. It should be noted that the heat pipe requires a starting period that may be longer than suitable for applications requiring a very short start-up, such as automotive (where current automotive specifications call for key-in to drive-off in 15 seconds). Freedom from such constraints in the typical heavy duty application allows the use of a remote heat source and a sodium heat pipe incorporated in the current design.

This offers three major advantages:

- Freedom to design the expansion heat exchanger to satisfy in an optimal fashion the heat and

aerodynamic requirements of the working fluid. Such is not the case in a direct flame heater where severe limitations are imposed on the heat exchanger design by the difficult heat transfer between the hot flue gasses and the heat exchanger tubes. Moreover, the heat exchanger is much simpler and easier to make as neither fins, other heat transfer augmentation devices, or special heater cage geometries are required.

- The ability to use different fuels (solid, liquid or gaseous) with replacement of the combustor alone.

- Uniform temperature of the heat exchanger enhances both the efficiency and reliability of the engine and eliminates the need for complex measures to achieve even distribution of flue gasses - a common problem of direct flame heaters. Even in a well designed direct flame heater, substantial temperature gradients appear on the heat exchanger wall. Furthermore, their locations, shapes and magnitudes are unpredictable. Thus the temperature control thermocouple will be located either in a "hot region" or in a "cold region" with adverse effects on efficiency or reliability and life respectively.

3.1.3 Working Fluids: Within the limitations of existing and near term technology the use of a heat pipe as the heat transport means in combination with deployment of metals for the expansion heat exchanger dictate the use of helium as the working medium. Though hydrogen engines will be characterized by specific power and/or efficiency higher than those of helium engines it is unknown at present to what extent the diffusion of hydrogen through the hot heat exchanger metal tubes interferes with the condensation process of the sodium vapor on the outer walls of the tubes. If it does not interfere, only a device to remove the hydrogen from the heat pipe is required. Such is within the scope of existing technology and can be a window made of material through which hydrogen will permeate but the sodium vapor will not (such as nickel or palladium alloys). Otherwise eliminating or substantially reducing the hydrogen diffusion by coating or by using ceramic heat exchangers and thus making it possible to employ hydrogen in remote heat source engines is one area where advanced technology offers potential improvements in the Stirling engine.

3.1.4 Materials: Solid glass ceramics cylinder and regenerator housing liners were incorporated in the conceptual design. They were designed to alleviate the need for creep resistant materials for the cylinders and regenerator housings. In addition, pistons made of a light weight, porous, non-metallic

material having low conductivity were incorporated in the design (STI is in the process of filing for a patent for pistons made of that material).

3.1.5 Relative Magnitude of Losses: Engine parameters and dimensions were selected to result in a particular balance of various losses that will yield the highest efficiency subject to the design requirements and constraints. Specific losses (such as flow losses in the regenerator and heat exchangers) were considered in relation to the other losses and an optimal balance achieved. A systematic search for the set of parameters and dimensions resulting in the highest efficiency, hereinafter refered to as "optimization for efficiency," was conducted using an N.V. Philips Gloeilampenfabrieken computer program.

3.2 CONCEPT DESCRIPTION

The considerations described in the preceding Section 3.1 were applied to a conceptual design of a Stirling engine rated 150 hp @ 1200 rpm and designated 4-720 HDSE and shown in figure 5.

Some specifications:

Arrangement: 4 double-acting cylinders symmetrically arranged about a common axis, one regenerator-heater-cooler assembly per cylinder.

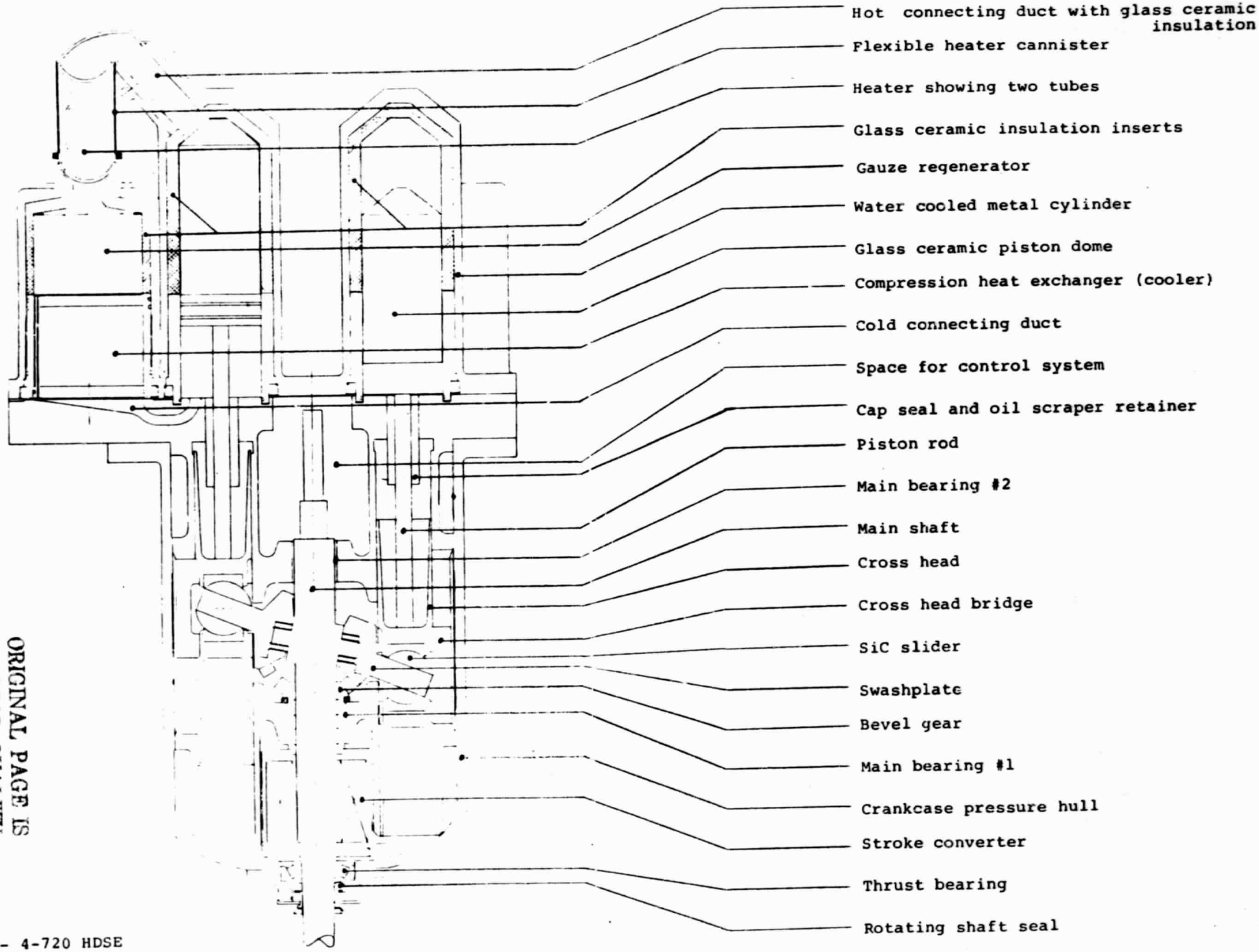


Figure 5 - 4-720 HDSE

Overall length: 1100 mm (44")

Cross sectional dimensions: 510 mm x 510 mm (20" x 20")

Cylinder bore: 100 mm

Maximum piston stroke: 91.4 mm (3.60") at 22° swashplate angle

Mean pressure: 11 MPa (1600 psi)

Working fluid: Helium

Heater temperature: 800°C (1472°F)

Cooler temperature: 45°C (113°F)

Regenerator diameter: 135 mm (5.32")

Regenerator length: 100 mm (3.94")

Regenerator structure: Gauze

Important components and design features are described in this section. Their principles, advantages and virtues are discussed. Included are the following features:

- Expansion heat exchanger
- Glass ceramic insulation
- Drive and power control

3.2.1 Expansion Heat Exchanger: A simple tube bundle comprising some 200 small diameter tubes about 5 inches in length form a very simple and efficient expansion heat exchanger. Compared to the usual heater cage of complicated geometry with fins and much larger tubes, the advantages of using a heat pipe become obvious.

The heat transfer from the condensing sodium to the tubes is independent of the tube geometry and arrangement, thus the simple tube bundle geometry. The diameter and length of the tubes are determined solely by the heat transfer properties from the tubes to the working gas. This is true since the external film coefficient is many times higher than the internal - much the opposite of direct flame heaters. Therefore, a large number of small tubes can be used. This brings about another advantage: due to the small tube diameter the material stress is sufficiently low to allow operating at 800°C even when non-strategic material such as AISI304 is used. This improves the engine efficiency while maintaining an adequate life.

The tube bundle is connected via a manifold to the hot connecting duct which is, in turn, bolted to the cylinder and sealed with a commercially available metal O ring.

This arrangement is suitable for mass production since only the small mass of the tubes and manifold need to be put in the brazing oven to be later connected to the cylinder. The traditional way of brazing each individual tube to the cylinder requires the large mass of the cylinder to reside in the brazing oven, which makes the brazing process slow, expensive and unsuitable for mass production.

The large number of small diameter tubes used here serves to reduce the total mass of the heat exchanger while maintaining adequate flow and heat transfer area.

Due to the sharp heat front that develops in the heat pipe at starting, some of the tubes will reach operating temperature while others are still cold. Therefore, flexibility has to be built into each individual tube. This is done by shaping the tubes in a wave form in a plane as shown in the figure. This automatically allows for the relative motion, due to thermal expansion, between the heat exchangers and the cylinder axes as well. The thin cannister surrounding the tube and containing the sodium is itself completely flexible. The hemispherical caps enclosing the heat exchanger serve to reduce the space between the heat exchanger and the regenerator and to accommodate tubes of equal length.

3.2.2 Glass Ceramic Insulation: The cylinder hot part, the regenerator housing and the hot connecting duct are made of glass ceramic insulation material shrink-fitted and bonded (glued or brazed) to water-cooled external metallic sheaths. Under cycle pressure the glass ceramic material is still under residual compression which it can well withstand along with the high temperature it experiences. The containing metal cylinders are in tension equilibrating the internal pressure in addition to the residual compression of the glass ceramic insulation. Being water-cooled, however, they will not experience creep and therefore may be made of simple and inexpensive material.

Only the transition parts on both ends of the heater, which are not water cooled, have to resist creep and high temperature. These parts, however, are of small diameter and may be made thin while still subjected to an acceptable level of pressure induced stress. They will then require only a small amount of creep resistant material and, furthermore, will not conduct much heat axially.

This way more than 80% of the heat and creep resistant material, usually containing strategic elements such as cobalt and nickel, commonly found in Stirling engines is eliminated. The 4-720 engine requires only 36 g/hp of creep resistant material - a factor of 5 less than the Philips/Ford 4-215 automotive engine, even though the latter, being a high speed engine (170 hp at 4000 rpm), has a very high specific power.

It is important to note that this scheme is only possible because glass ceramics have practically no thermal expansion and therefore there is no problem with incompatible thermal expansion between the ceramic inserts and the water-cooled external sheaths.

Philips has measured average thermal expansion values of 9×10^{-7} in the temperature range of room to 900°C. This is a factor of 20 less than the characteristic expansion of metals.

This scheme is not suitable for very small engines since

the insulation thickness, for equal performance, is independent of the engine size and in order to fit it in small engines the cylinders will have to be moved apart, with an adverse effect on the mechanical losses.

For large engines, such as the 4-720 HDSE the insulation thickness that will not require the cylinders to be moved apart and influence the mechanical efficiency is such that no major improvement in efficiency is brought about: the radial heat conduction through the insulation is of the same order of magnitude as the axial heat conduction through the metal walls it replaced.

This, however, is based on available solid glass ceramics with thermal conductivity of 0.6 $w/m^{o}C$. A major improvement in efficiency will be brought about by using porous glass ceramics with much lower thermal conductivity as described in the following section 4.1.

An additional advantage of the glass ceramic insulation is that it eliminates thermal stresses in the regenerator housing and the cylinder. These components, when made of metal in the normal fashion are subjected to axial thermal gradients which induce thermal stresses in them. This, very often, poses a limitation on their design which prevents optimal efficiency. This problem is more severe in the regenerator design: very often mechanical design, taking into account the thermal stress in the regenerator housing will yield a longer regenerator, characterized by

higher flow resistance than would be necessary.

Obviously water-cooled metal regenerator housing with insulation inserts of extremely small coefficient of thermal expansion will develop no thermal stresses, allowing the regenerator to be as short and wide as is called for by the thermodynamic and aerodynamic requirements of the cycle. In this sense it may be said that even the solid glass ceramic insulation enhances the efficiency of the engine.

3.2.3 Drive and Power Control: The drive design was motivated by the wish to reduce contact forces and hence enhance its efficiency, life and reliability. The resulting design is compact, rigid and characterized by low mechanical friction.

The maximum angle of the swashplate is 22°. It is varied by rotating the swashplate relative to the tilted section of the main shaft. This rotation is affected by the two-vane hydraulic stroke converter and transmitted to the swashplate via a bevel gear attached to the stroke converter housing and mounted to the shaft with needle bearings. The control mechanism (shown in figure 6) comprising a pressure pump, spool valve governor and its actuating linear stepper motor, fits in the crankcase between the cross-head cylinders on the cycle side. Hydraulic fluid is supplied to and returned from the stroke converter via concentric tunnels in the main shaft. The

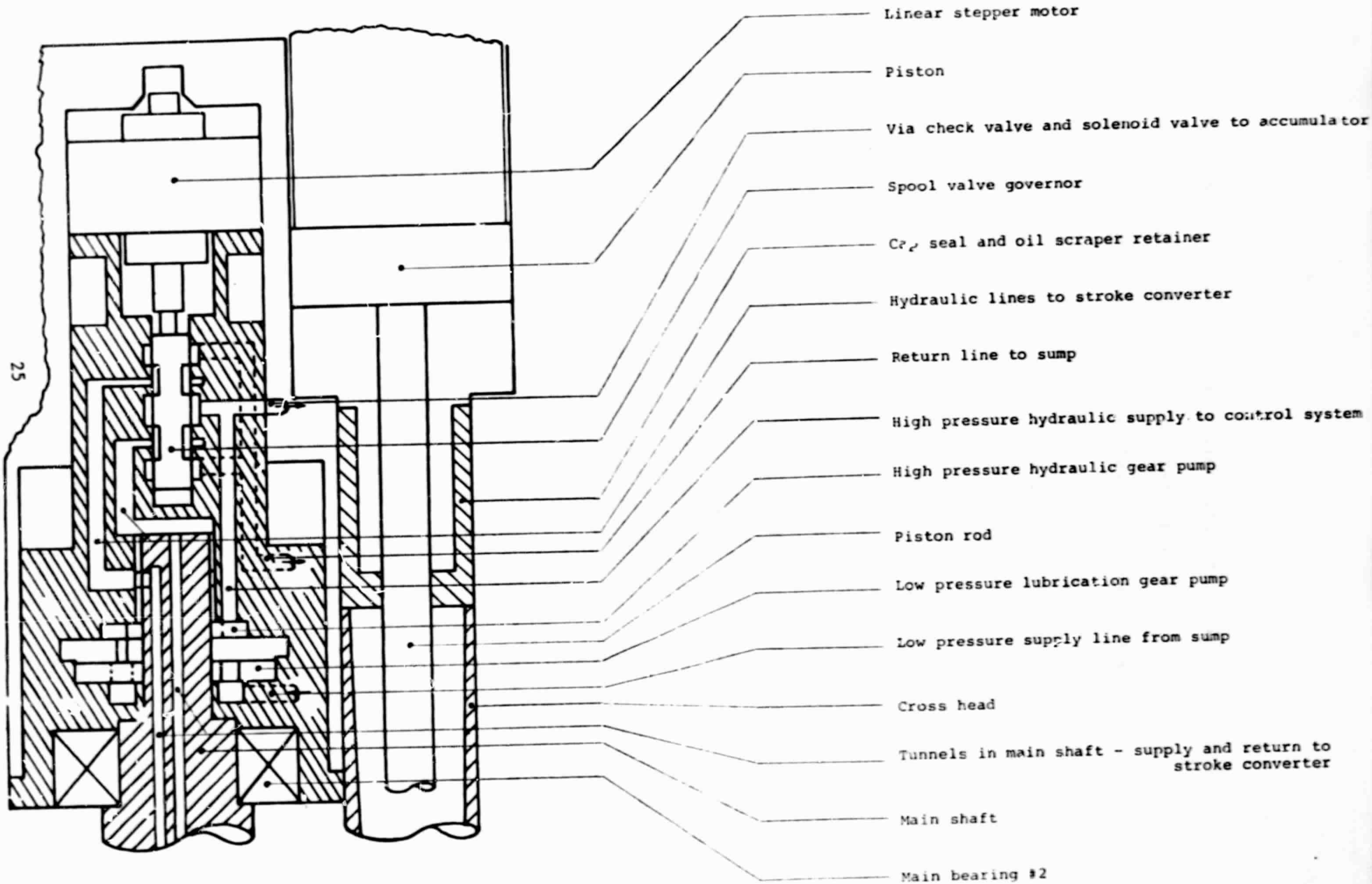


Figure 6 - Power control system

lubrication pump is integrated into the control package and may communicate with an external dry sump reservoir.

The main bearing's distance is made very short in order to prevent excessive shaft flexture. This is done by mounting one of the main bearings on the bevel gear sleeve. The main bearings are moderately loaded hydrodynamic plane bearings.

The cross-heads are long, in order to reduce contact forces, and are very simple and suitable for mass production. Each cross head is made of two identical, rotationally symmetric parts shrink-fitted into the relatively short and rigid bridge section. The sliders are made of silicon carbide which helps to reduce their weight and to prevent fretting.

The compact drive and power control fits into a small barrel-shaped crankcase which is maintained at cycle pressure. This makes it possible to avoid the reciprocating shaft sealing problem.

A single, commercially available rotating shaft seal, designed to be in oil at all times, contains the working fluid. Cap seals on the piston rods isolate the crankcase from the cycle pressure variation and metal scrapers on the piston rods prevent oil migration into the working space.

The compactness of the drive, while enhancing the mechanical efficiency and the strength of the drive components, gave rise to some difficulty in the dynamic balancing of the drive. The rotating inertia was not sufficient in order to balance the reciprocating

mass. In order to rectify the situation the swashplate had to be weighted with tungsten, which adds about $100 to the cost of the engine.

This allows for perfect dynamic balance at 19° with residual unbalanced moment causing vibration with maximum amplitude of only 20μm (figure 7).

Piston ring friction was reduced about 60% by using a single piston ring on each piston instead of the usual system of two rings with the extreme cycle pressure between them. This improves the mechanical efficiency by about 30% but requires some positive action to be taken in order to counter the pumping action of the piston rings which will tend to cause unequal pressure in the four cycles.

This is accomplished by using two cap seals mounted close to each other and making a small axial groove in the piston rod positioned in such a way that when the piston is at mid-stroke the cycle will be connected via the groove with the space between the cap seals called the intermediate plenum.

Figures 8a and 8b schematically illustrate this arrangement.

On the down stroke the axial groove (1) in the piston rod faces the top cap seal (2) when the piston is at mid-stroke, connecting the intermediate plenum (3) with the cycle. The pressure in the intermediate plenum then becomes equal

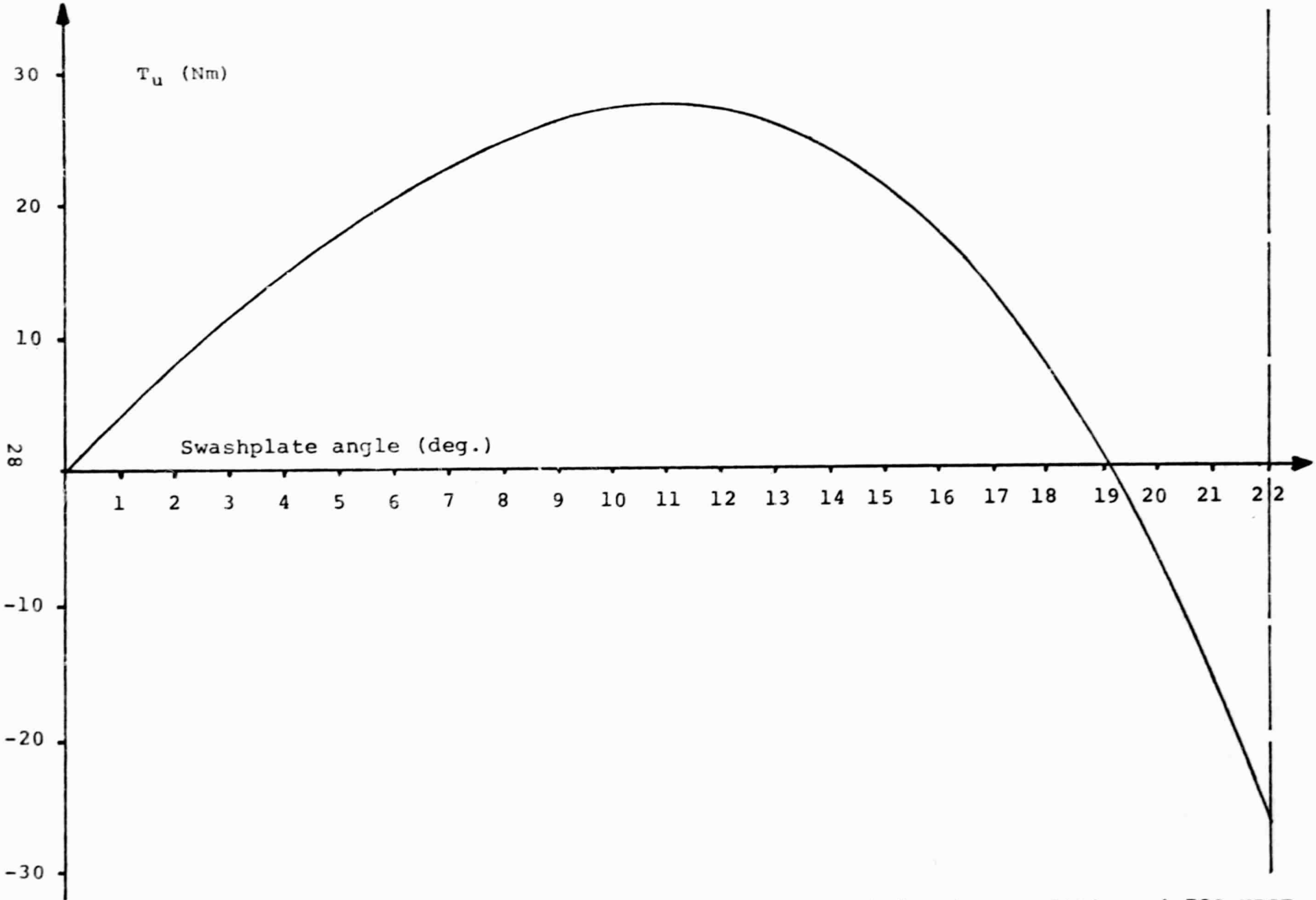


Figure 7 - Unbalanced inertial torque (T_u) for optimum balancing condition, 4-720 HDSE

to the cycle pressure at that point (P_{md} in figure 8c). A short time later the groove faces the bottom cap seal (4) connecting the intermediate plenum with the main plenum (5) causing the pressure of the main plenum to equal that of the intermediate plenum which is P_{md}. On the up stroke when the groove connects the two plenums nothing happens since at that point they are at equal pressure P_{md}.

A moment later the groove will connect the intermediate plenum with the cycle causing the plenum pressure to change to P_{mu} (in figure 8c). The main plenum pressure stays at P_{md}. This arrangement ensures that the main plenum pressure is P_{md} so long as the stroke is larger than the distance between the two cap seals $X_s - X_m$.

The four main plenums (one for each cycle) are interconnected forcing all cycles to have the same pressure P_{md} when their respective pistons are at mid-position on the down stroke.

A capillary tube, connecting the crankcase with one of the cycles, compensates for gas leakage through the oil scraper.

Such leakage, though very small, is inevitable since P_{md} is smaller than the mean cycle pressure present in the crankcase.

This arrangement was never used before on any Stirling engine since the idea was not conceived until

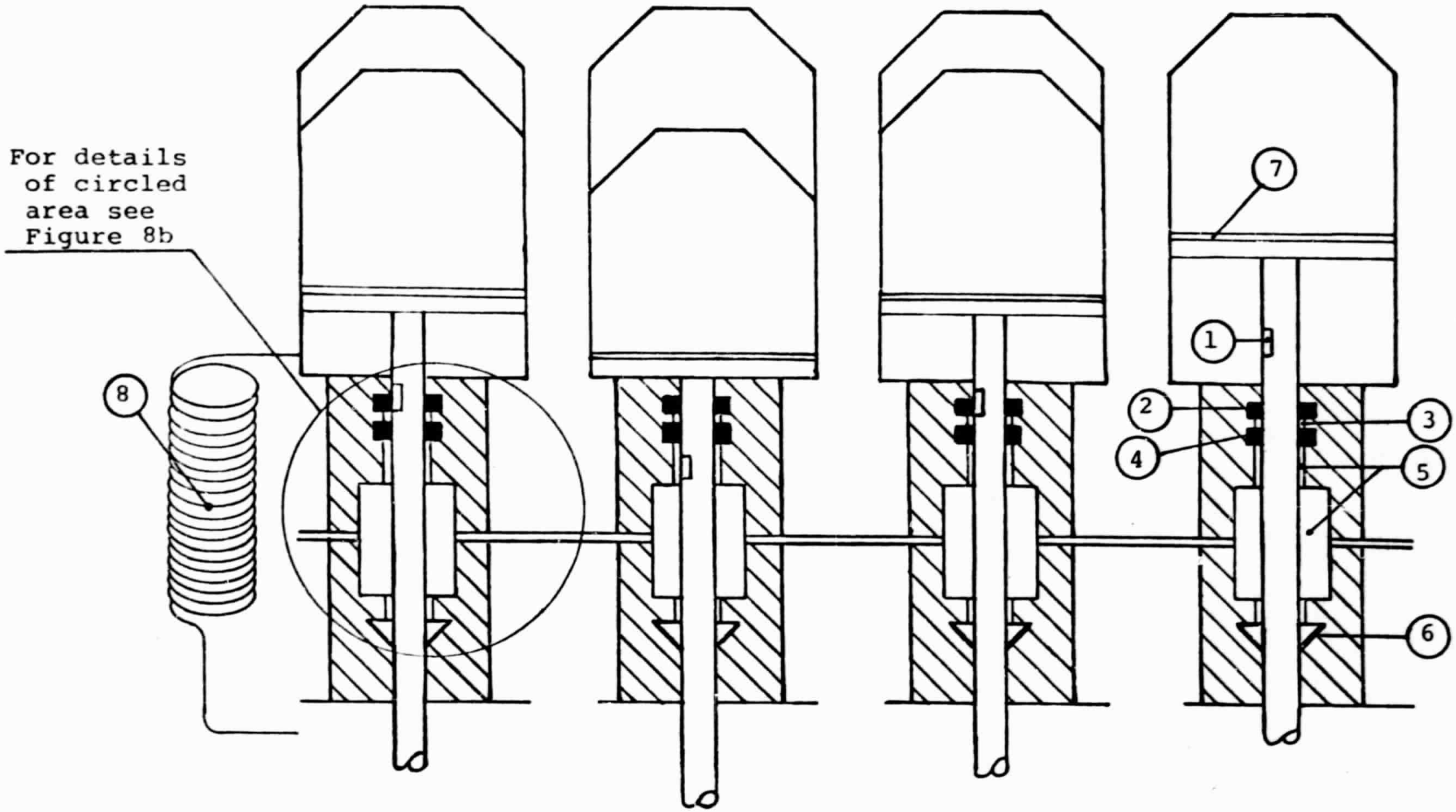


Figure 8a - Pressure function equalizing arrangement

1) Groove
2) Top cap seal
3) Intermediate plenum
5) Main plenum
6) Oil scraper
7) Piston ring
8) Capillary tube

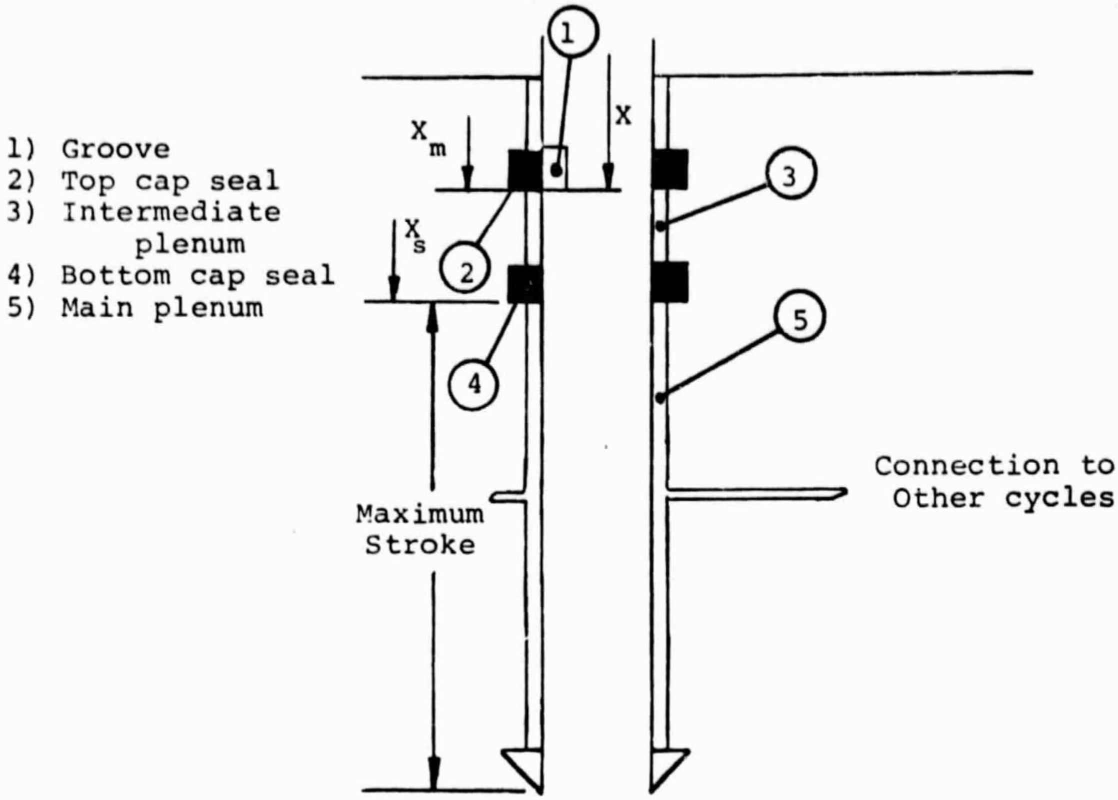


Figure 8b

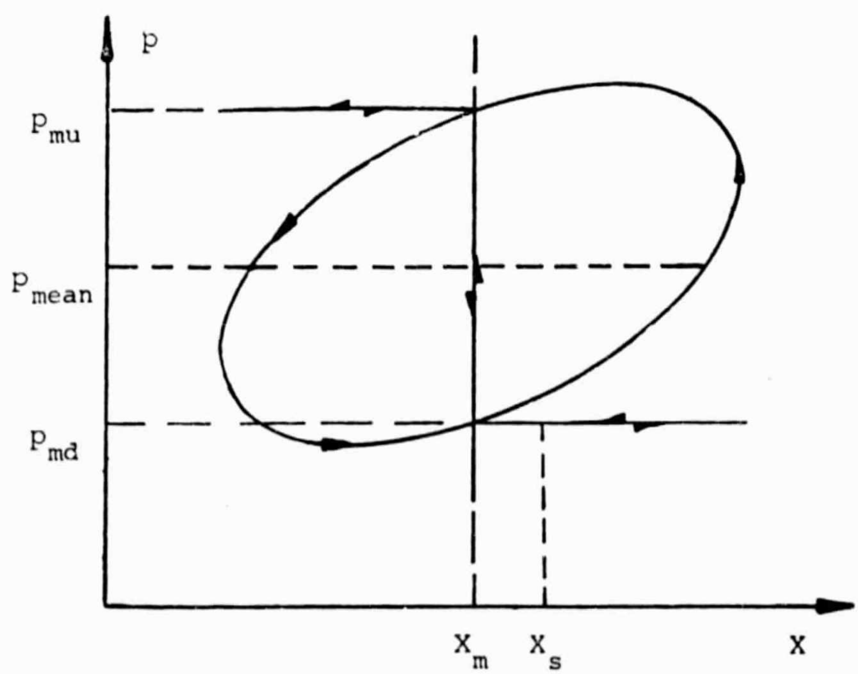


Figure 8c

recently.

3.3 EXPECTED PERFORMANCE

The 4-720 HDSE described in the preceeding Section 3.2 was simulated at 16 operating points with torque and speed values ranging from idle conditions to maximum. Simulation was done using Philips' data validated computer program described in Appendix A. The resulting performance is summarized in the engine map, power curves and torque curves shown in figures 9, 10 and 11 respectively, as well as in table 1.

The maximum indicated efficiency is 52% at 1200 rpm and maximum swashplate angle (22^{o}). This is the "design point" yielding 165 shaft horsepower with no auxiliaries. It was chosen at 1200 rpm to be compatible with a 3 pole alternator for such applications that require electric AC power at 60 Hz.

Mechanical efficiency is very high as explained in Section 3.2.3 and also because the indicated efficiency is high. Mechanical efficiency is a monotonically increasing function of the indicated efficiency.

Indicated efficiency is proportional to the sine of the phase angle between the pressure and volume variations. Hence higher indicated efficiency is equivalent to larger

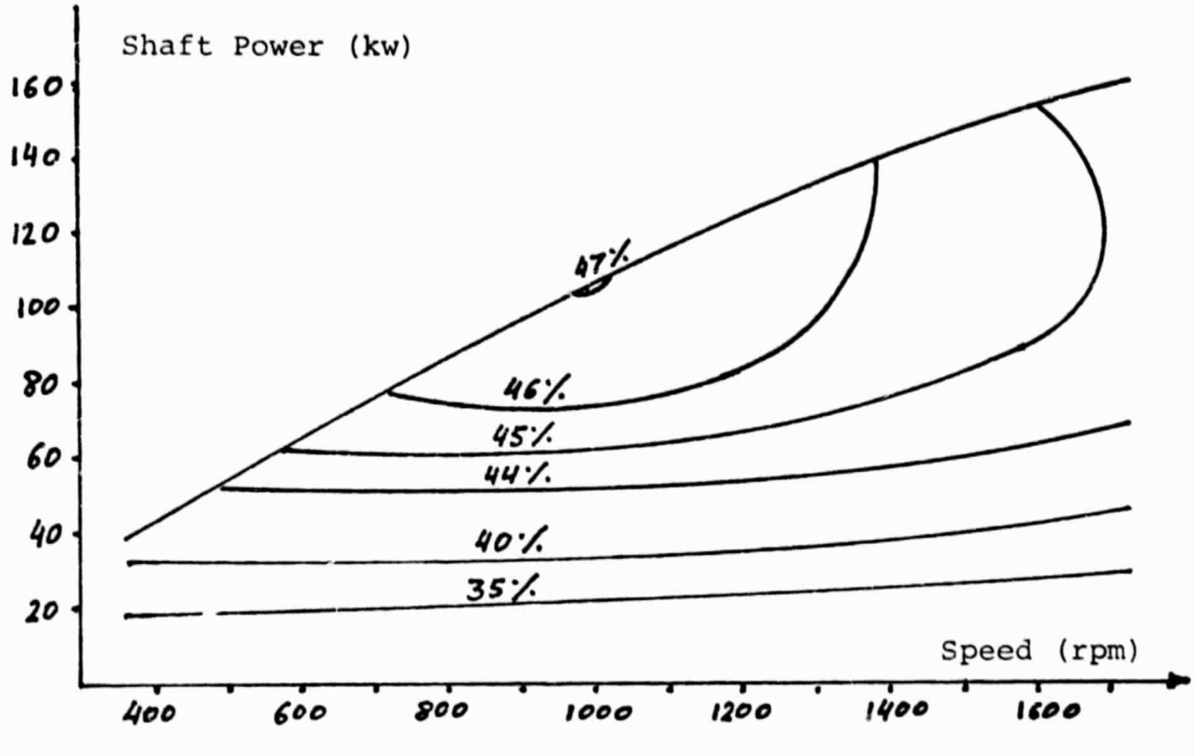


Figure 9 - 4-720 Engine map showing lines of constant shaft efficiency

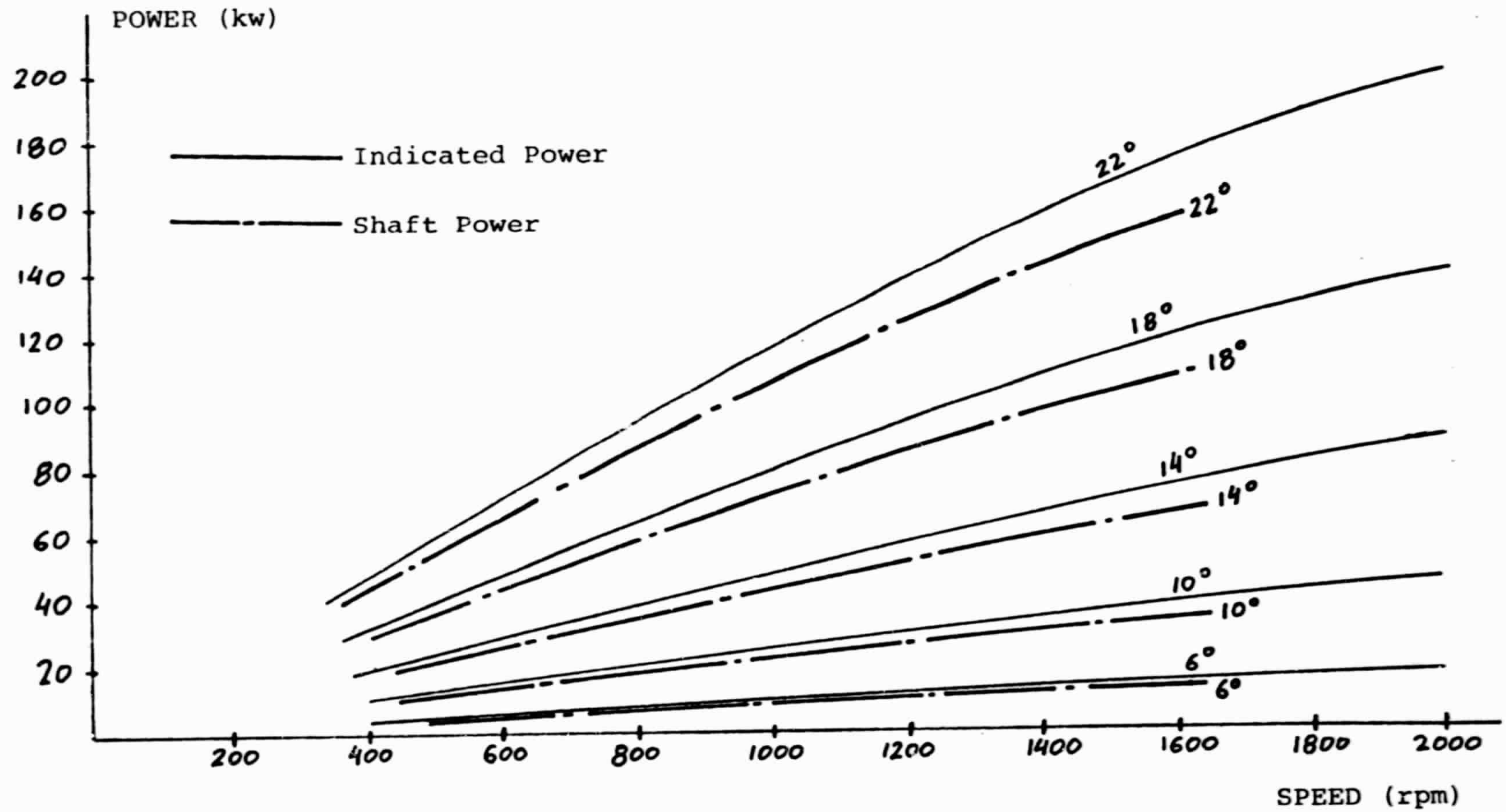


Figure 10 - Indicated and shaft power vs. speed for various swash plate angles for 4-720 HDSE

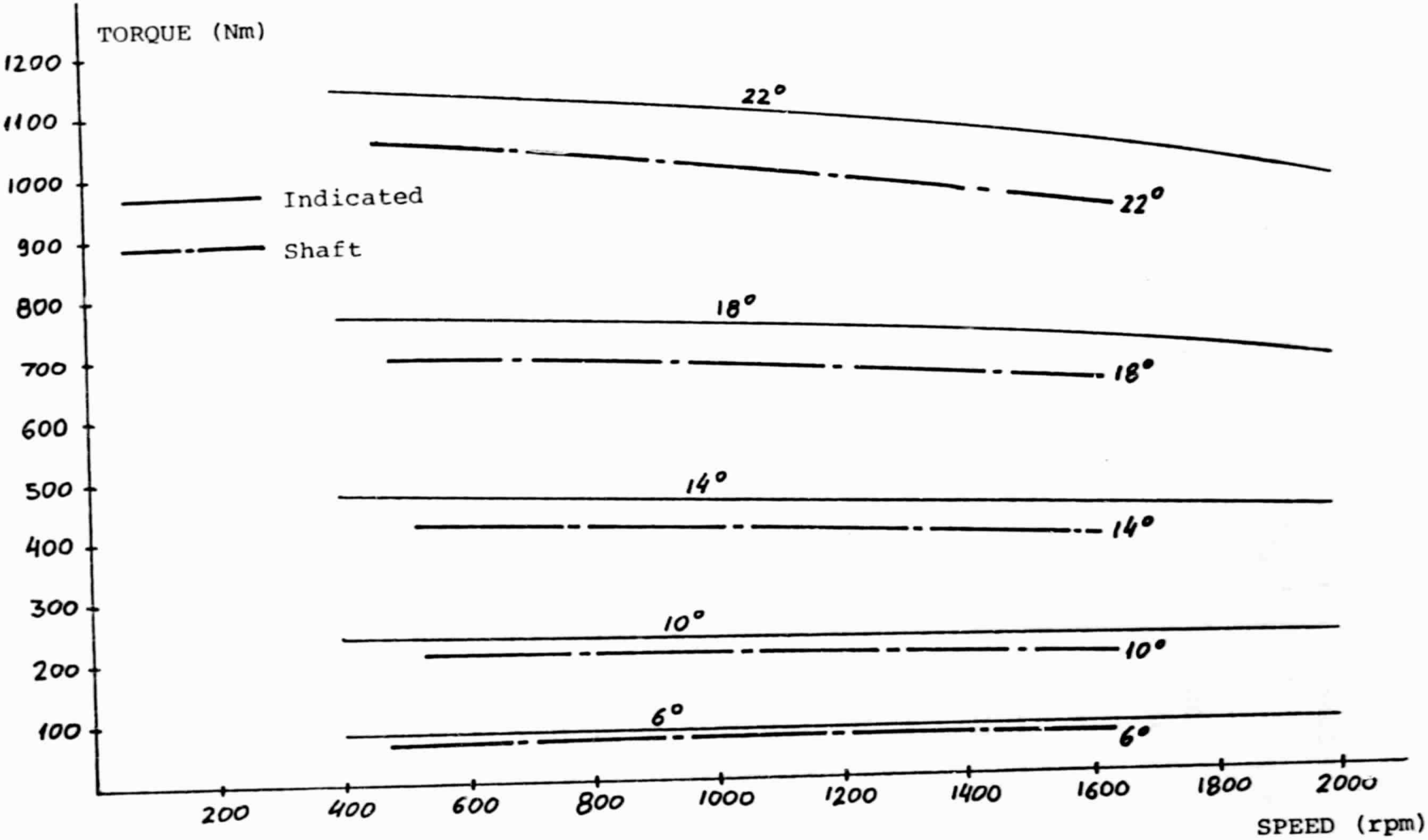


Figure 11 - Indicated and shaft torque vs. speed for various swash plate angles for 4-720 HDSE

Swashplate angle (deg)	Speed (rpm)	Indicated power (hp)	Shaft power (hp)	Indicated efficiency (%)	Shaft efficiency (%)
22	1600	232	207	50.6	45.2
	1200	184	167	51.3	46.6
	800	126	116	50.6	46.5
	400	64	60	45.8	42.6
18	1600	160	142	51.0	45.3
	1200	125	113	50.8	45.9
	800	84	77	49.0	44.7
	400	43	40	42.4	39.5
14	1600	98	86	49.9	43.7
	1200	76	68	48.3	43.3
	800	52	47	45.1	40.8
	400	27	24	36.5	33.3
10	1600	51	44	44.6	38.7
	1200	39	34	42.0	36.8
	800	27	24	37.1	33.0
	400	13	12	27.2	24.2
6	1600	19	16	31.4	26.0
	1200	13	11	27.6	22.6
	800	9	8	22.2	18.7
	400	5	5	14.0	11.9

<u>Table 1</u> **4-720 HDSE Performance Data** Using Solid Ceramic Cylinder and Regenerator Liners; Pistons Made From Porous Ceramic.

phase angle.

The ratio of the forces in the direction of motion ("power producing forces") in the drive mechanism to forces normal to the direction of motion ("friction producing reaction forces") is also roughly proportional to the phase angle between the pressure and volume variations.

This ratio, multiplied by the average friction coefficient is the mechanical efficiency which is thus a monotonic function of the indicated efficiency.

Consequently, the maximum shaft efficiency is 47% in the neighborhood of the design point.

Although a design point was selected and the engine optimized thereat, the 4-720 HDSE is by no means a "single operating point engine." As can be seen in the brake engine map (figure 9) the efficiency hardly varies with engine speed and varies only slightly between full power and about a third thereof. This will make the engine suitable for many different specific applications and result in exceptionally low energy usage regardless of the specific duty cycle.

3.4 INFLUENCE OF ENGINE SIZE

As was mentioned in Section 2 above, heavy duty engines range in size from a few kilowatts to a few megawatts. Even though 70% of these engines are under

200 hp and 90% under 350 hp it is still very important to find out whether engines larger than the 4-720's 150 hp also offer the high efficiency potential demonstrated for the 4-720 HDSE.

In order to address this question a 500 hp engine, designated 4-1350 HDSE was conceptually designed. It shares the same configuration with the 4-720 HDSE and is shown in figure 12.

Some specifications:

Arrangement: 4 double-acting cylinders symmetrically arranged about a common axis, one regenerator-heater-cooler assembly per cylinder

Overall length: 1500 mm (59")

Cross sectional dimensions 860 mm x 860 mm (34" x 34")

Cylinder bore: 173 mm (6.81")

Maximum piston stroke: 134 mm (5.28") at 22° swashplate angle

Mean pressure: 11 MPa (1600 psi)

Working fluid: Helium

Heater temperature: 800°C (1472°F)

Cooler temperature: 45°C (113°F)

Regenerator diameter: 248 mm (9.76")

Regenerator length: 100 mm (3.94")

Regenerator structure: Gauze

Maximum Speed 1500 rpm

Design Speed 900 rpm

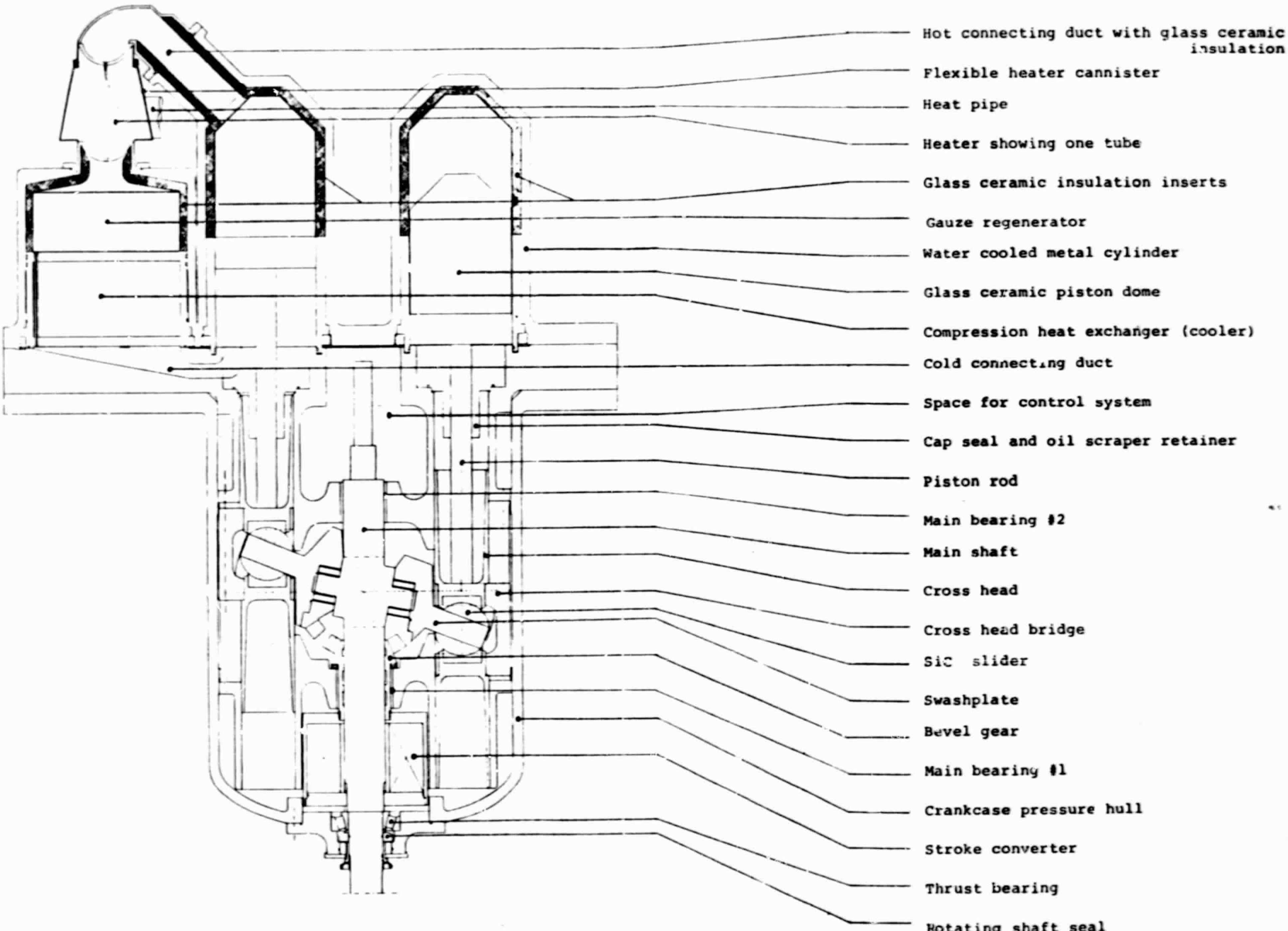


Figure 12 - 4-1350 HDSE

The engine was simulated and its predicted indicated performance is summarized in figures 13, 14 and 15 and in table 2.

The simulation was done with porous ceramic insulation with thermal conductivity of 0.2 $w/m^{0}C$ and hence, the maximum indicated efficiency of 56% should be compared with the corresponding value for the 4-720 HDSE with porous ceramic insulation, 54%.

The difference is due to the fact that the 4-1350 engine has a smaller cylinder and regenerator surface area per unit swept volume than the 4-720 engine. Since both have the same insulation thickness the relative heat conduction loss of the 4-1350 engine is smaller than that of the 4-720 engine. This signifies a basic trend: glass ceramic insulation inserts are more effective for larger engines.

In general the 4-1350 HDSE exhibits the same potential and the same advantages as the 4-720 engine. The choice of 150 hp for the conceptual design, therefore, does not have any significant influence on the results of this study.

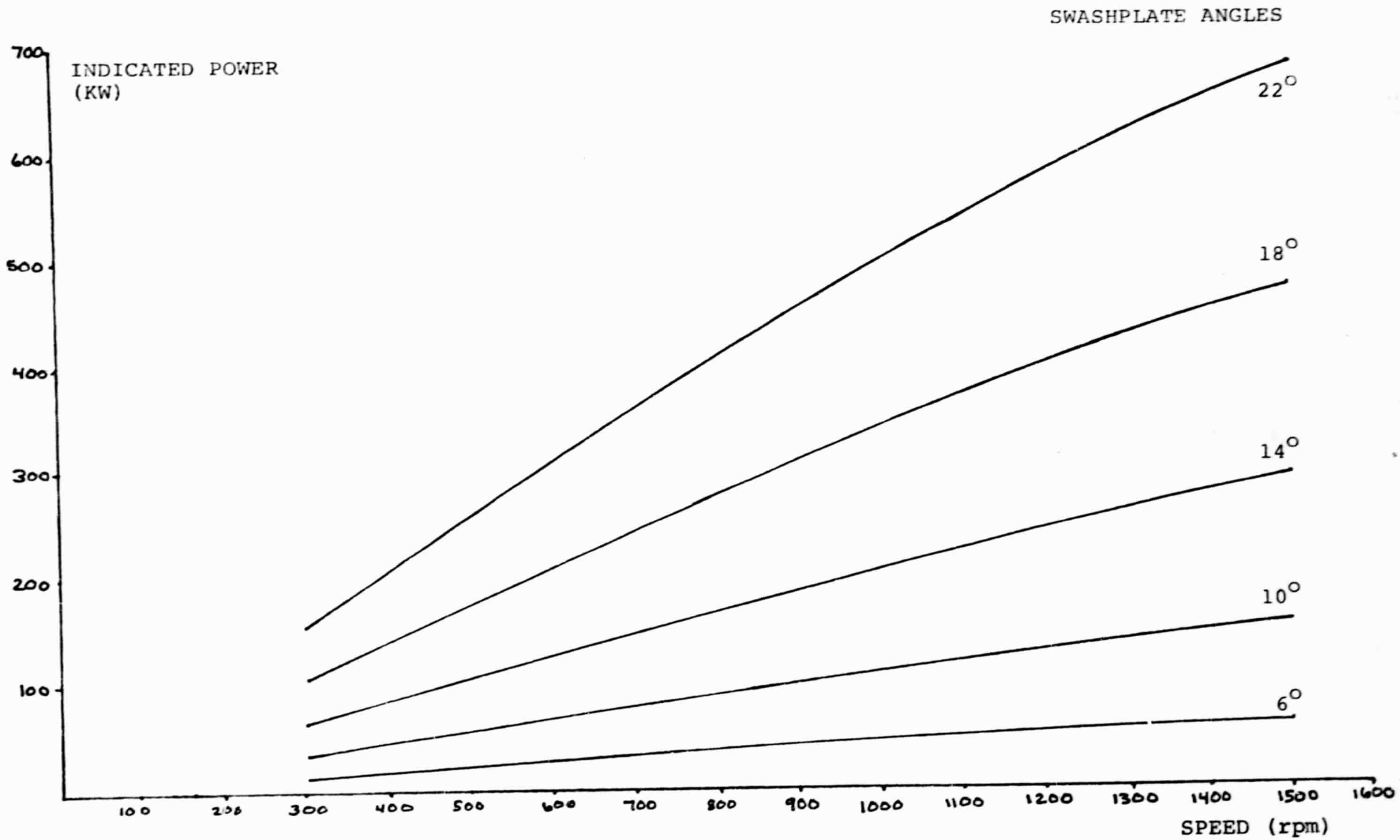


Figure 13 - Indicated power vs. speed for various swashplate angles for the 4-1350 HDSE

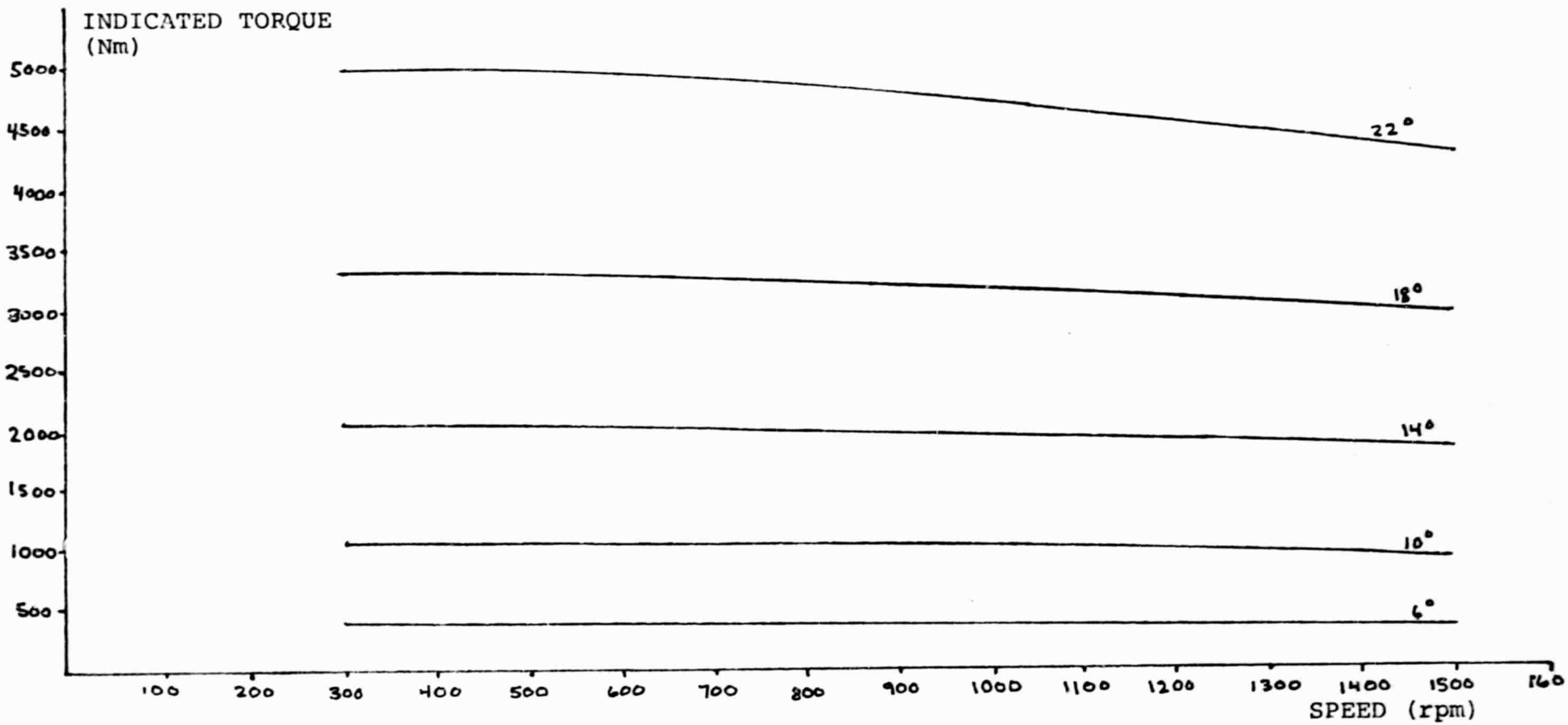


Figure 14 - Indicated torque vs. speed for various swashplate angles for the 4-1350 HDSE

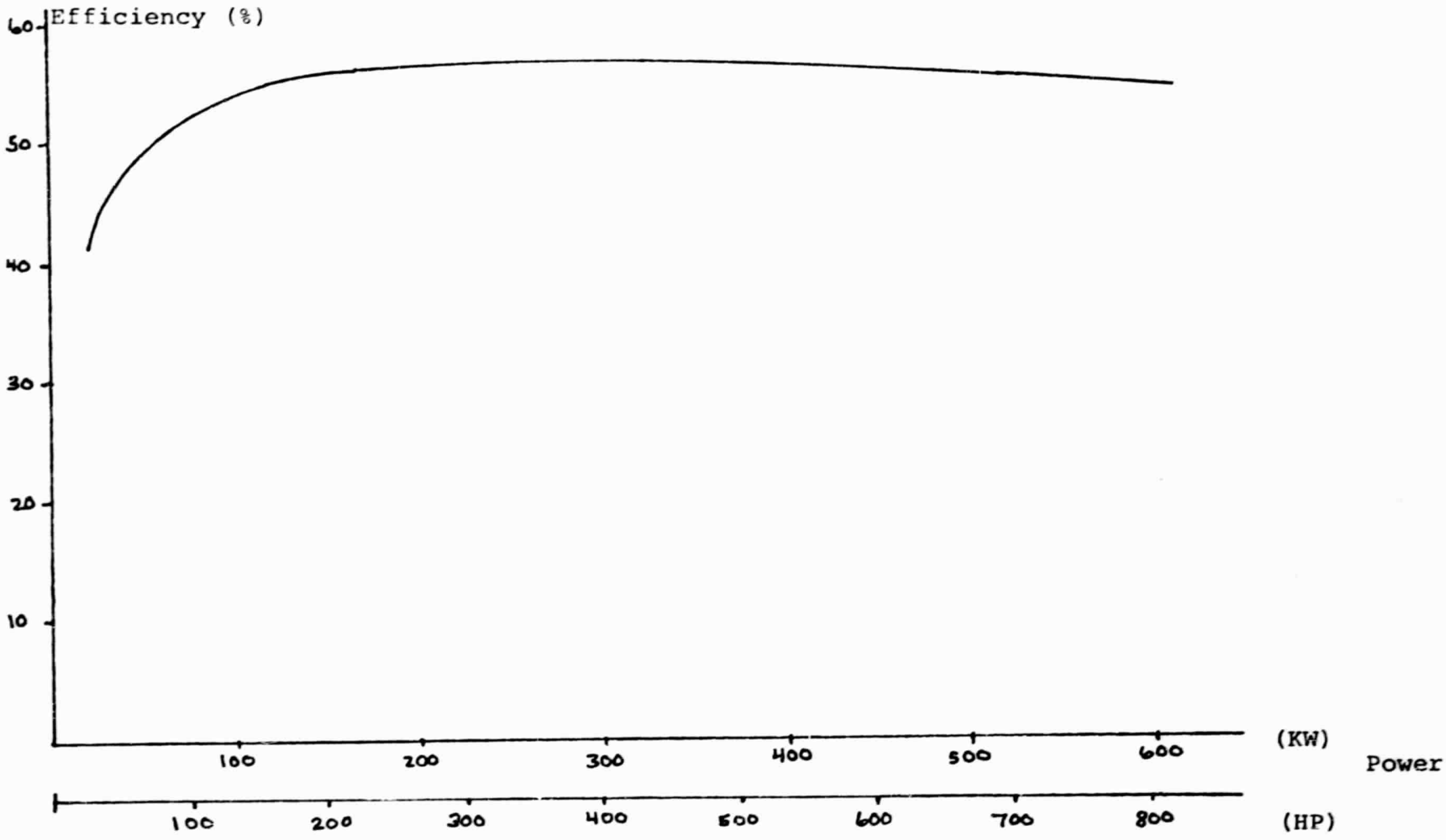


Figure 15 - Indicated efficiency curve at design speed of 900 rpm for the 4-1350 HDSE

Swashplate angle (deg)	Speed (rpm)	Indicated power (hp)	Indicated efficiency (%)
	1500	908	53.0
	1200	767	54.7
22	900	601	55.7
	600	414	55.7
	300	212	53.5
	1500	627	54.9
	1200	524	56.0
18	900	406	56.4
	600	279	55.9
	300	143	52.6
	1500	390	56.0
	1200	323	56.4
14	900	249	56.2
	600	170	54.8
	300	87	49.9
	1500	205	55.4
	1200	168	55.1
10	900	129	53.9
	600	88	51.1
	300	45	43.6
	1500	75	49.7
	1200	61	48.0
6	900	47	45.1
	600	32	40.0
	300	16	29.7

Table 2 4-1350 Performance Data

Section 4

APPLICATION OF ADVANCED TECHNOLOGY

Further improvements of the HDSE may be brought about by the application of technology that is not readily available and requires some development work. Three areas where advanced technology may be applied were investigated. They are:

- Deployment of porous, closed-cell glass ceramic insulation
- High expansion heat-exchanger temperature
- Using hydrogen as the working fluid while maintaining remote combustion with a heat pipe

The degree of difficulty in applying these technologies varies with the specific item under consideration and for the second item (high expansion temperature) also with the extent to which the temperature is raised.

Potential improvements associated with the technologies investigated were assessed both separately and coupled and are discussed in this section. Conclusions regarding the relative balance of effort and potential improvement of these technology items are presented in Section 5.

4.1 POROUS, CLOSED-CELL, GLASS CERAMICS

It was pointed out in Section 3.2.2 above that

deployment of state of the art, solid glass ceramic insulation, while drastically reducing the engine cost, will have only little effect on its efficiency. This stems from that fact that if, in order to reduce the radial heat conduction, the insulation thickness is increased, requiring cylinders which are spaced farther apart and resulting in additional mechanical losses in the drive system from increased moments on the crossheads. These losses will largely cancel out the gain from the reduced heat conduction. If, however, different material with lower thermal conductivity were used, heat conduction will be reduced with no effect on the mechanical losses and thus an improvement in engine efficiency, particularly at part load where heat conduction plays a major role, will be brought about.

Obviously the material used should be heat resistant and characterized by extremely low coefficient of thermal expansion. Such material could be porous glass ceramics, provided that its structure is such that would not allow the working gas to shuttle in and out of the pores, as this would result in some losses. Closed cell, or at least substantially so, porous glass ceramic that may be produced by sintering hollow nucleated-glass spheres will be suitable for this application. It is conservatively estimated that thermal conductivity value of at least 0.2 w/m^{o}C can be achieved with this material.

The 4-720 HDSE, with insulation having thermal conductivity value of 0.2 w/m^{o}C, was simulated (at the

design speed of 1200 rpm and load varying from full load to 10% thereof. Indicated and shaft efficiency were plotted versus power and compared to the equivalent quantities for the engine with solid glass ceramics (0.6 w/m$^{\circ}$C). The resulting improvements in indicated and shaft efficiency are shown in figures 16 and 17 respectively.

4.2 HIGH EXPANSION TEMPERATURE

In order to determine the effect of the heater temperature on the indicated efficiency of the HDSE, four different engines maintaining the swept volume, general configuration and power output of the 4-720 HDSE were optimized to yield the highest efficiency each at a different temperature ranging from 1500°F to 2500°F.

This procedure was done twice: first for helium working fluid and then for hydrogen. Two corresponding curves showing efficiency at full power versus the heater temperature summarize this effort and are shown in figure 18. Note that every point in each of the curves represents a slightly different engine. They all share the same configuration, swept volume and power output but they are slightly different from each other so that each yields the highest efficiency at the temperatures at which it operates and with its working fluid.

There is an improvement in efficiency with higher

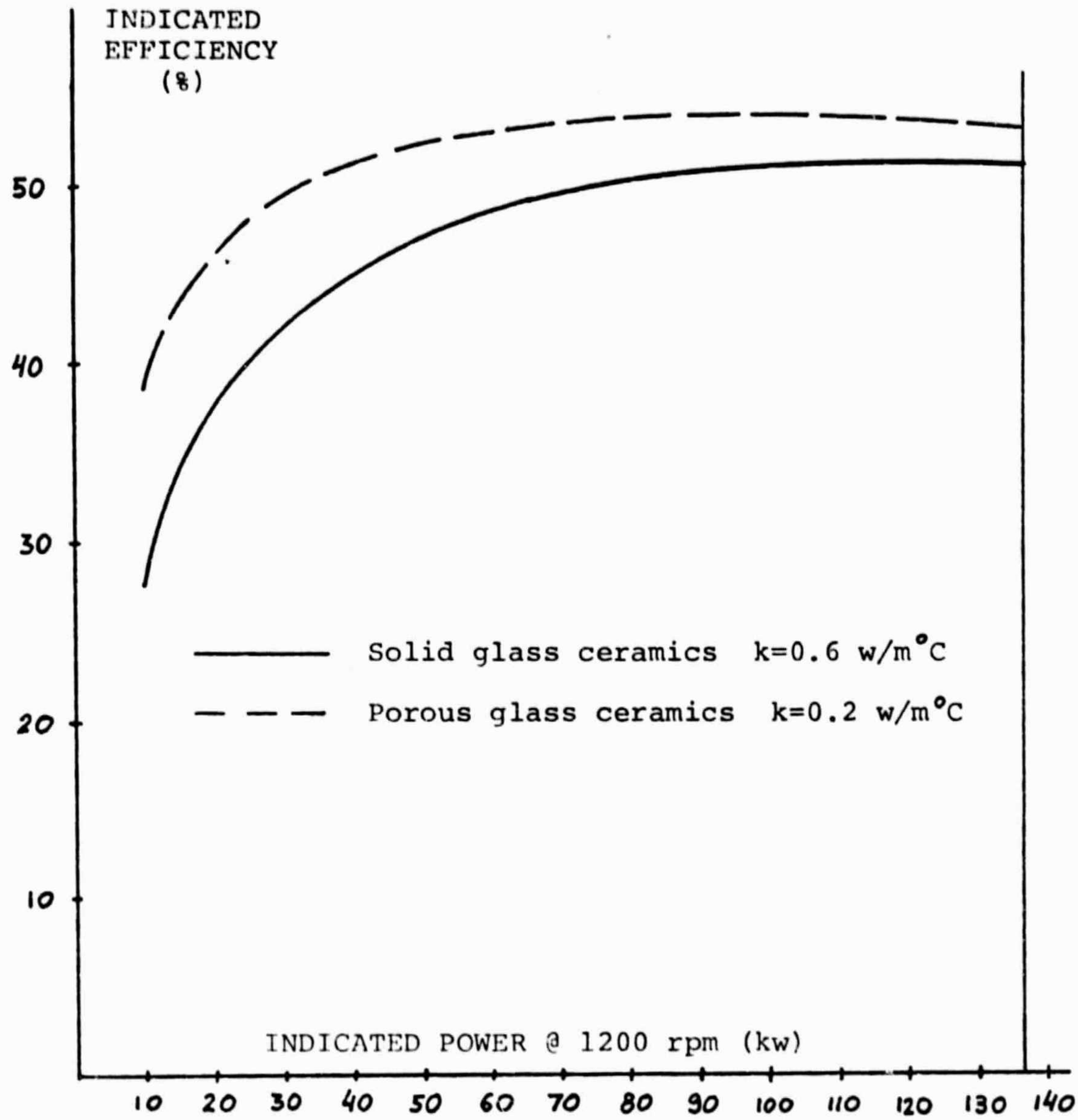


Figure 16 - Efficiency improvement potential of closed-cell porous glass ceramic inserts with low heat conductivity (k)

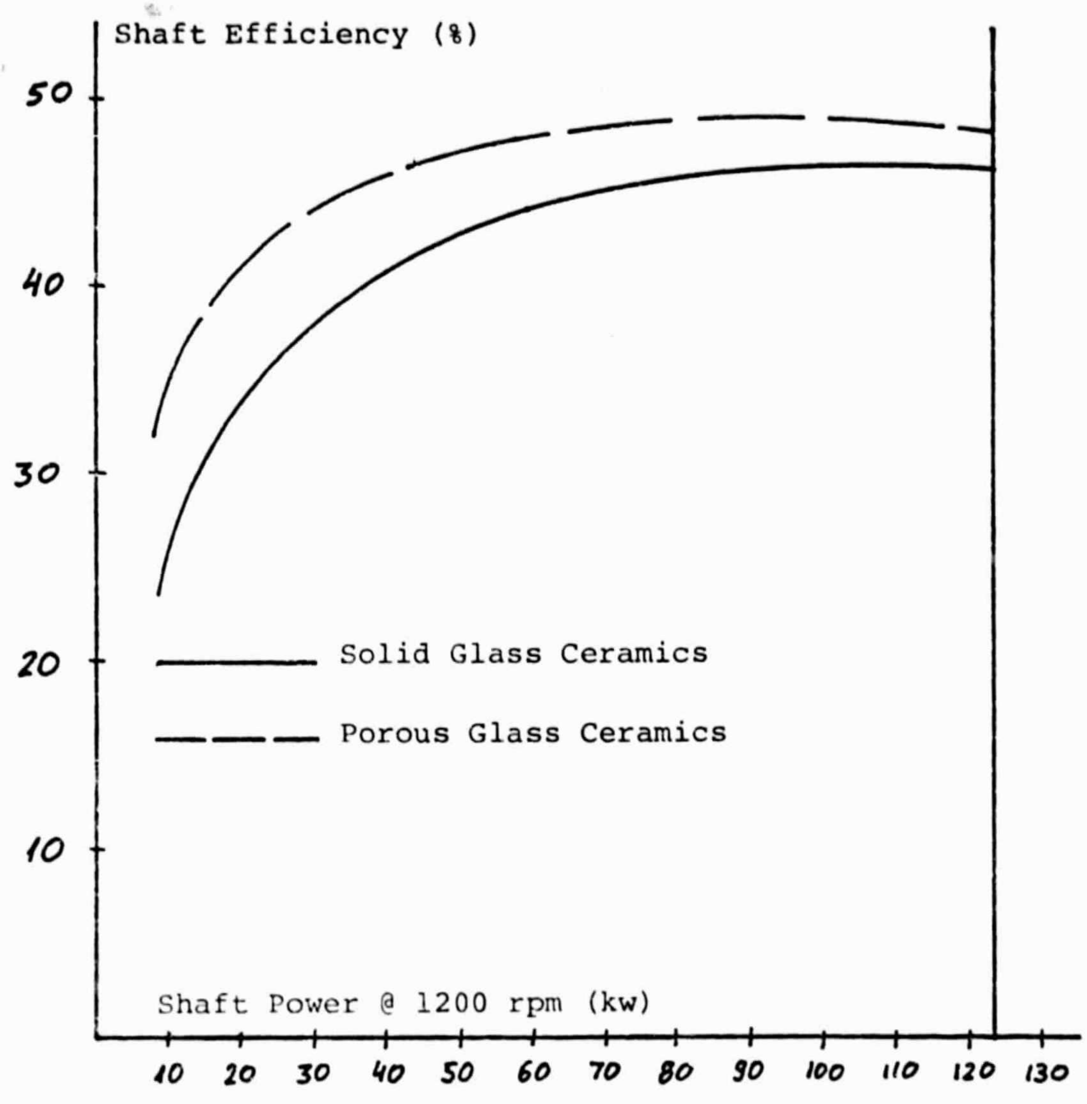


Figure 17 - Part load shaft efficiency at 1200 rpm

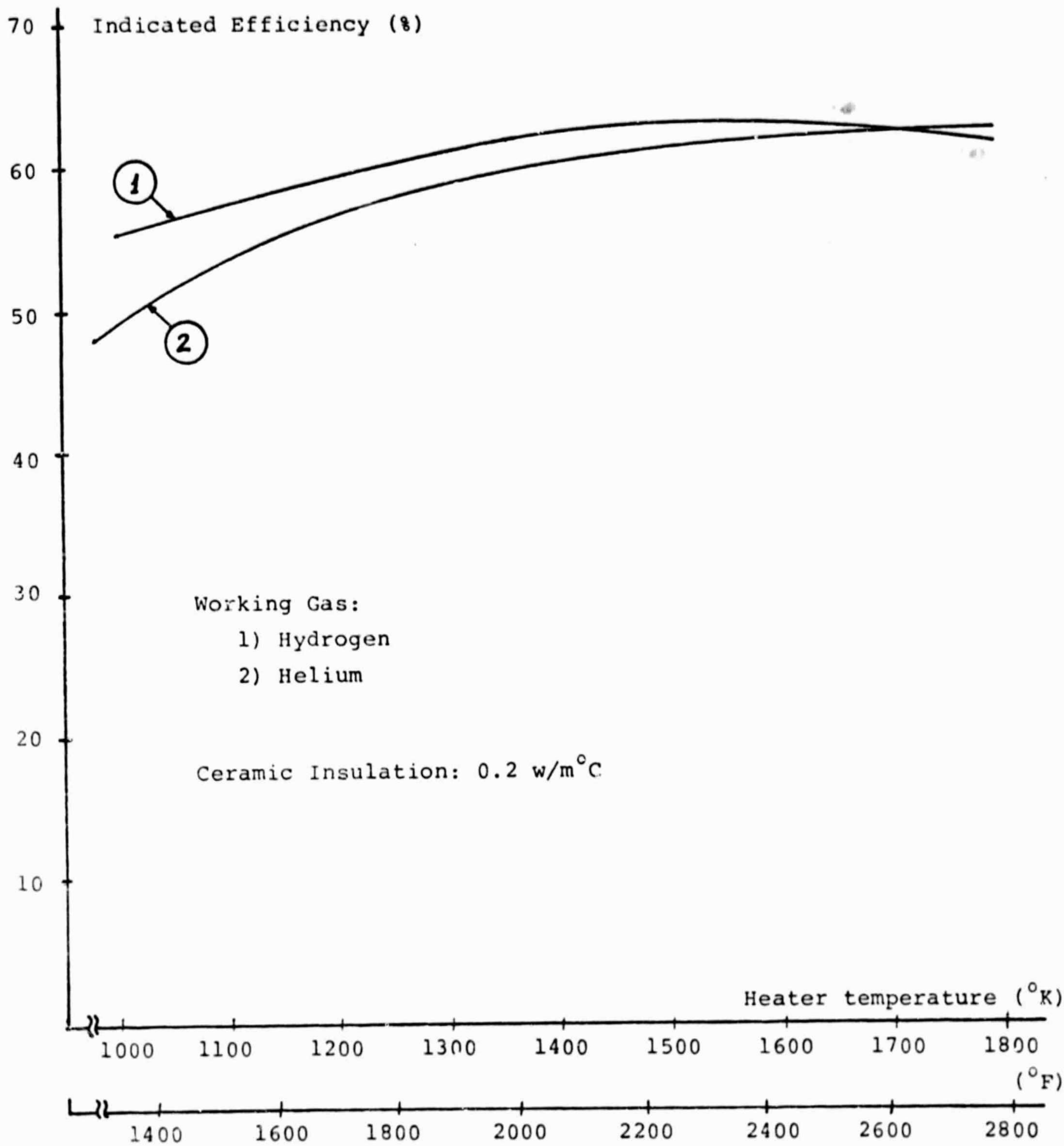


Figure 18 - Dependence of indicated efficiency on heater temperature at constant power output and engine swept volume

heater temperature which becomes less pronounced as the temperature goes up. At a certain high temperature level no further improvement may be achieved by raising the temperature still higher. In fact, efficiency will drop if this is done. The point where the curve levels off occurs at a lower temperature for hydrogen (2200°F) than for helium (2700°F).

The explanation for this behavior is that heat conduction increases more rapidly than the semi-adiabatic efficiency ($1-T_c/T_h$) as the heater temperature goes up so that after a while whatever gain is due to the semi-adiabatic efficiency is lost to increased heat conduction. Heat conduction losses are due to heat leaks through the engine hardware as well as through the working fluid. The latter brings about regeneration losses that are very important and accounts for the difference in behavior between helium and hydrogen. The thermal conductivity of hydrogen at high temperatures is much higher than that of helium and goes up with temperature much faster than that of helium, as shown in figure 19.

4.3 HYDROGEN WORKING FLUID

The results of the effort described in the previous section 4.2 and shown in figure 18, are used to assess the improvements due to using hydrogen.

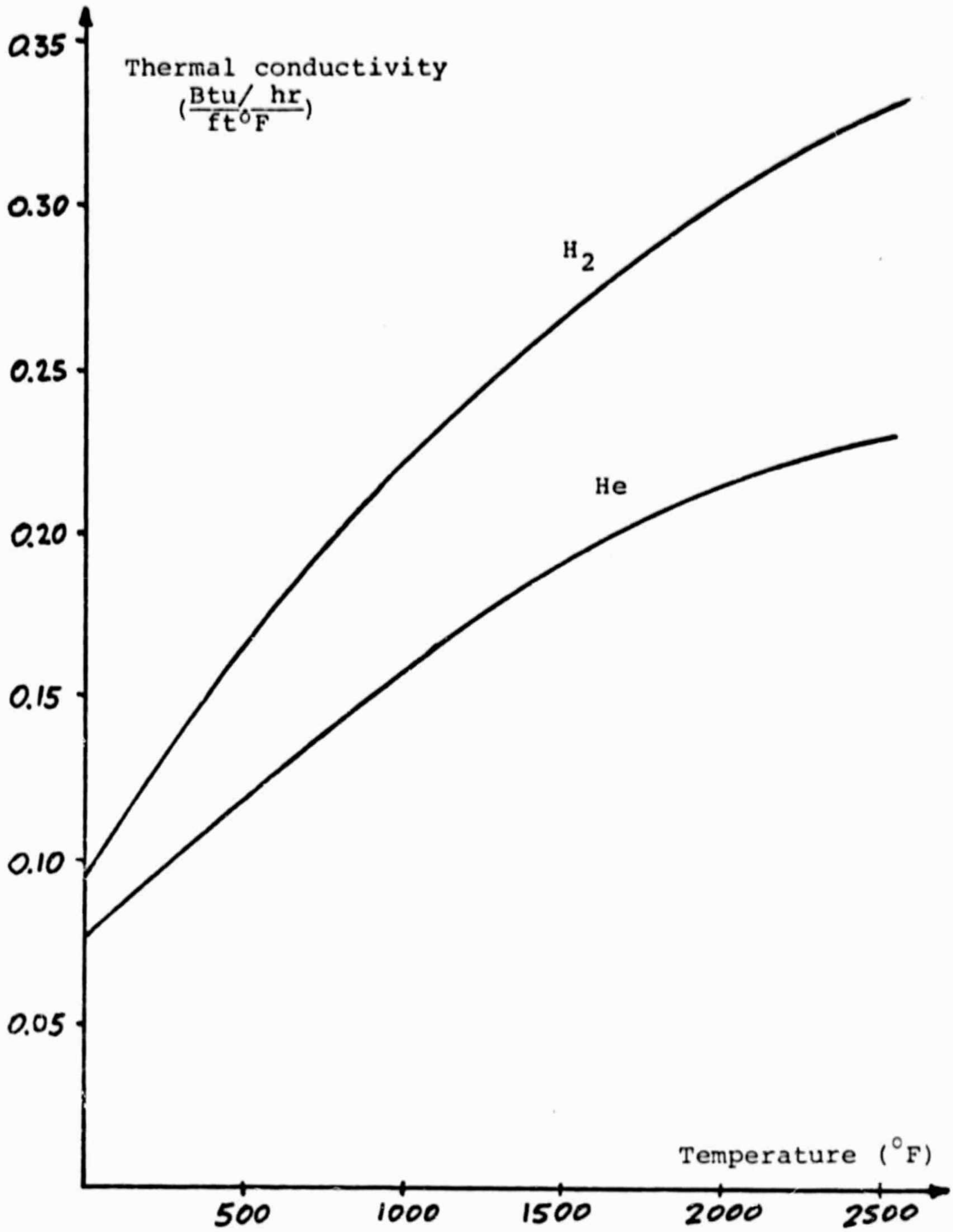


Figure 19 - Thermal conductivity of helium and hydrogen

In general, hydrogen causes an improvement in the efficiency of the engine. This is due to the fact that being less dense and having lower viscosity than helium it produces less flow losses. Also, being composed of diatomic molecules, its heat capacity ratio is 1.4, lower than the monoatomic helium's 1.67 and hence hydrogen will reduce the cycle adiabatic losses.

With the current state of the art engines characterized by metal heat exchangers and a temperature level of 1400°F this improvement is rather pronounced: about 15% (or 7 points). As seen in figure 19, however, the rapidly increasing thermal conductivity of hydrogen completely changes the picture at higher temperatures where heat conduction through the hydrogen will add so much more regeneration losses than helium at higher temperatures as to offset its other advantages. In fact, above 2600°F higher efficiency will be achieved with helium than with hydrogen.

Section 5

CONCLUSIONS

The Stirling engine, when properly designed, will compete effectively in the heavy duty engine marketplace. The 4-720 HDSE, representative of an engine designed for this market, has all the features required for it to compete with I.C. engines. These include long service life, low maintenance expectation, resistance to adverse environmental conditions (hermetically sealed) and weight similar to heavy duty diesel engines. At the same time its design is very simple and suitable for production in quantity at low cost. Material cost is also low since more than 80% of the creep and heat resistant material usually needed for the Stirling engine was eliminated by using glass ceramic insulation inserts in the cylinders, regenerator housings and hot connecting ducts. The heat resistant material that is still required for the engine does not contain any strategic elements. It is estimated that when mass-produced it should not be more expensive than a comparable diesel engine.

The 4-720 HDSE is superior to the diesel engine in a number of aspects. First and formost, the HDSE has fuel flexibility; it may even use such fuels as biomass and powdered coal. This feature is becoming increasingly important with the high cost of gasoline and diesel fuel.

Furthermore, the performance of the HDSE is far superior to that of the diesel engine, particularly at part load. Compared to hydrogen automotive Stirling engines, the efficiency of the existing technology HDSE is, in general, not higher, but is achieved with far lower complexity and potential for cost and much higher reliability and longer life.

Application of advanced technology will increase the efficiency of the HDSE more than the automotive engine since the most promising item - porous glass ceramic insulation - is less suitable for the latter for lack of space.

Additional advantages of the HDSE over the comparable diesel are quiet, vibration-free operation and low exhaust emission levels produced by its combustor (not included in this study).

Of the advanced technology items that were investigated under this effort the most promising and "effort effective" is deployment of porous, closed-cell glass ceramic insulation inserts. Efficiency improvements of at least 36% at part load and 4% at full load may be brought about with relatively little effort by applying this technology.

The other two items are using hydrogen as the working fluid while maintaining the heat pipe, and raising the heater temperature. It was determined that the former is much more effective than the latter. At the current technology level, which limits the heater temperature to about

1500° F, switching from helium to hydrogen will raise the full load efficiency by the same amount as raising the heater temperature 200° F. This is even more pronounced at part load, where the additional heat conduction in the working fluid due to higher heater temperature plays an important role.

Raising the heater temperature even 200° F requires use of a ceramic heater which, it seems, is more difficult than the effort required in using hydrogen with a heat pipe.

APPENDIX A —

PHILIPS STIRLING SIMULATIO" CODE

The Philips Stirling cycle simulation code was compiled (for manual calculation) in 1950 and has been continuously refined and modified. Today it is the most advanced, accurate and at the same time relatively simple code in existance. It is a mixture of "second" and "third order" numerical and analytical methods. Processes the interaction of which cannot be neglected are treated as a coupled system whereas some processes are decoupled and superimposed on the coupled system if their influence on other processes can be neglected. Of the decoupled processes, some are treated analytically when the approximations necessary for analytical (temporally non-discrete) treatment are negligible.

A wealth of experimental data was used by Philips to determine what approximations are permissible. Numerous experiments were also performed on many different Stirling engine configurations in order to construct empirical methods for processes that are difficult to treat analytically. Such empirical formulae are then backed up theoretically and the various coefficients they incorporate are truly global and applicable to any

Stirling cycle machine of any size and configuration. To illustrate this point it is worthwhile to mention that even gas temperature fluctuations within the engine, a very difficult quantity to measure, were measured by Philips in order to gain insight into the unsteady heat transfer in various engine parts.

This carefully weighed mixture of analytical, numerical second and third order methods and the wealth of empirical results incorporated in the analysis is the key to the accuracy and reliability of the Philips simulation code as well as its relative simplicity and computational efficiency.

Derivatives of Philips Simulation Code

The existance of the simulation code described above together with the availability of extensive volume of experimental results enabled Philips, with good understanding of the cycle to construct a simpler, purely analytical (non-numerical) simulation code and accurately determine its correlation to the numerical code described above. Correlation factors of global nature were incorporated in the analytical code to yield a less accurate code, sufficiently reliable and extremely fast and efficient. This code has been incorporated in a synthesis procedure used as a design tool. This procedure searches in the

engine parameter superspace for an optimum point (e.g. highest efficiency or lowest specific weight, etc.) subject to a number of constraints posed as required performance or other values (e.g. power output or total length, etc.). The result of this procedure is a set of engine parameters yielding the optimal engine satisfying the design requirements. Obviously this procedure executes a cycle simulation for every combination of the engine parameters requiring hundreds of simulations to provide the optimal choice. It utilizes the analytical simulation procedure which requires very little computation work and is still sufficiently accurate to render this synthesis tool feasible and very economical to use.